A Modified Oil Lubrication System with Flow Control to Reduce Crankshaft Bearing Friction in a Litre 4 Cylinder Diesel Engine

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Paul J. Shayler, Li Cheng, and Qile Li
University of Nottingham

Emad Wahab
Ford Motor Company

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Abstract

The oil distribution system of an automotive light duty engine typically has an oil pump mechanically driven through the front-endancillaries-drive or directly off the crankshaft. Delivery pressure is regulated by a relief valve to provide an oil gallery pressure of typically 3 to 4 bar absolute at fully-warm engine running conditions. Electrification of the oil pump drive is one way to decouple pump delivery from engine speed, but this does not alter the flow distribution between parts of the engine requiring lubrication. Here, the behaviour and benefits of a system with an electrically driven, fixed displacement pump and a distributor providing control over flow to crankshaft main bearings and big end bearings is examined. The aim has been to demonstrate that by controlling flow to these bearings, without changing flow to other parts of the engine, significant reductions in engine friction can be achieved. The study has been conducted on a 1.5litre, 4 cylinder turbocharged diesel engine. By reducing the feed pressure to the bearings from a baseline pressure of 3bar absolute to 1.5 bar absolute, reductions in engine rubbing friction mean effective pressure of up to 14% has been achieved at light load. Similar reductions in friction were recorded across a speed range of 1000-2000 rev/min and net indicated mean effective pressures up to 3.5 bar. The ranges were conservatively limited to protect against bearing damage. The paper reports details of the oil system modifications and the test results. The fuel economy benefit due solely to the friction reduction, not including any benefit from a reduction in oil pump work, is around 1½ % over the New European Drive Cycle (NEDC). The reduction in friction is demonstrably significant and represents an area with great potential to improve engine efficiency.

Introduction

The net indicated work done by the cylinder gases on the pistons of an internal combustion engine is greater than the brake work output. A fraction of the indicated work is consumed driving ancillaries and the valve train and a further fraction is dissipated by rubbing friction at the various interfaces between surfaces moving relative to each other. This can entail sliding motion such as between the piston and the cylinder liner, rolling-sliding motion between cams and roller followers, and rotational motion in the journal bearings. Collectively these produce a friction mean effective pressure, FMEP, of the order of 1 bar [1]. This is a significant part of the indicated load, given brake mean effective pressures under mixed European driving conditions are typically a few bar, reducing engine efficiency and causing engine wear.

Friction and lubrication in engines has long been the subject of tribological research [2] and efforts to improve engine efficiency and durability [3]. Tear-down friction studies of diesel engines show that the piston assembly and interaction with the liner is the largest contributor to friction losses, followed by the crankshaft assembly and the valve train [1]. Friction reduction in power cylinders has received considerable attention [4, 5] as have the development of low viscosity oils [6] and improvements in engine thermal management designed to reduce the penalty associated with higher friction during cold engine operation [7]. The higher friction is largely attributable to higher values of oil viscosity at low temperatures, and various routes to accelerate the rate of rise of bulk oil temperature after cold start-up have been explored. These include reducing effective sump capacity [8] or the introduction of baffles to direct hottest available oil to the oil pump inlet [9], external heating of the oil [10] and the use of thermal insulation [11]. The crankshaft main bearings are particularly slow to warm up because of the strong thermal coupling to surrounding metal in the bearing support plates and the large thermal capacity of the crankshaft. The engine structure in the lower part of the engine warms slowly because of low rates of heat conduction from areas around the cylinders which are heated more directly by heat transfer from the combustion system and exhaust gases. After a cold start, there is a rapid rise of several degrees in the oil film temperature of the bearings associated with frictional heating but the rate of temperature increase then slows to be similar to that of the immediate surroundings [12]. Introducing a high contact resistance at the outside surface of the bearing shells reduces heat transfer in this direction [13] but not into the journal, which limits the benefit.

The dual-pressure oil delivery system described in [14] supplies oil at low pressure up to mid-range loads and speeds, and at a higher pressure at high speeds and loads. The advantage of running at low pressure was the reduction in excess oil flow and pumping work. The principal reason for raising the oil pressure was to open the piston cooling jets. At any given operating condition, all the oil was supplied at either the low or high pressure. In the work reported here, the potential of a different approach to reducing friction in the crankshaft main bearings and connecting rod big end bearings has been investigated. In this approach, low pressure oil is supplied only to an oil gallery feeding the crankshaft main bearings, big end bearings and piston cooling jets whilst continuing to supply oil at a higher pressure to the valve train and turbocharger. The investigation was carried out on a firing engine after the required modifications were made to the oil system. These allowed the influence of reducing the pressure of the oil feed to the main and big end bearings to be explored without also changing the oil feed conditions to other parts of the engine. The aim was to reduce friction by restricting oil supply to the bearings. Generally, the bearings operate fully-flooded, with the lubricant filling the full width and circumference of the bearing. Previous studies suggest that bearing friction can be reduced by reducing oil feed pressure [15, 16]. The reduction is at least partly explained by observations of the film in a journal bearing with a transparent bush which show that when the oil supply is restricted, cavitation develops and the film area is reduced [17, 18, 19].

In a conventional lubrication system, the oil pump is mechanically driven by a pulley or gear drive off the crankshaft making the pump speed proportional to engine speed. The delivery pressure is usually measured in the main gallery and regulated by a relief valve or by adjusting the delivery per revolution of the pump. The gallery pressure is typically ~4 bar absolute under fully warm operating conditions, and significantly higher under cold running conditions. The coupling to engine speed and high delivery pressure means the oil pump makes a substantial contribution to parasitic work load on the engine. Electrification of the oil pump drive is one way to decouple pump delivery from engine speed [20, 21, 22] with the potential to reduce parasitic work, and the development of 48V pumps [23] makes this increasingly practicable. In the study reported here, which has been carried out on a test bed, the oil pump was driven by an electric motor powered externally, eliminating this source of parasitic work and allowing the pump speed to be varied with no dependence on engine operating condition. This does not directly affect the distribution of oil to the various parts of the engine which require lubrication or a minimum pressure for actuators in variable valve timing and lift systems. The distribution of oil flow from the main gallery is dictated by the flow resistances in each of the pathways to a nominal atmospheric pressure. In the modified system described later, the oil pressure of the feed to the valve train and turbocharger is controlled to match that of the unmodified system. The pressure of the feed to main and big end bearings and piston cooling jets is controlled independently.

Thus, in summary, the engine modifications allow the oil pump to be driven externally with independent speed control to vary oil delivery. The oil pump work does not contribute to the ancillary load on the engine. Oil delivery is split into a two paths. The pressure of the feed to the valve train and the turbocharger was held at a fixed 3.3-3.5 bar absolute (barA) for all tests. The second stream feed oil to the main and big end bearings and the piston cooling jets. The tests cover cases when the piston cooling jets are open and closed, strongly indicating that changes in engine FMEP can be associated solely with changes in the friction contribution of the bearings.

Experimental Setup

Engine Specifications

The test engine is a 4 cylinder, 1.5 litre turbocharged diesel engine (see Figure 1); the specifications of the engine are shown in Table 1. The engine was instrumented with pressure transducers and thermocouples to monitor temperatures and pressures of oil, coolant, air and exhaust gas. The encoder TDC was aligned with the cylinder-1 TDC using an AVL OT-SENSOR 428 tool set. National Instruments LabVIEW was installed to provide data acquisition. The data acquisition rate was set to 100Hz, and results were averaged over 100 data points, for steady-state test conditions.

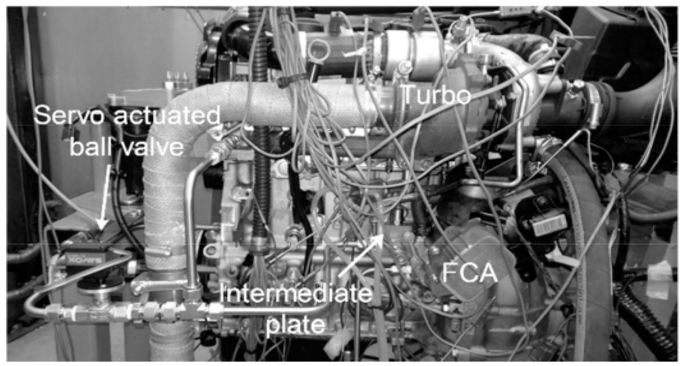

Figure 1. Test bed illustration photo.

Table 1. Engine specifications.

Engine type	In-line, 4-cylinder, turbocharged
Displaced volume	1560cc
Stroke	88mm
Bore	75 mm
Connecting Rod	136.8 mm
Compression ratio	17:1
Number of Valves	8
Number of main bearings	5
Number of big end bearings	4
Main bearing diameter	50mm
Main bearing length	20mm
Big end bearing diameter	47mm
Big end bearing length	16mm

A predicted breakdown of friction contributions made by the main component groups is given in Figure 2, for an engine speed of 1500 rpm and SAE 5W-30 oil at a bulk temperature of 90°C. The predictions were made using the model described in [1]. The oil pump contribution has been deleted from the ancillaries because in the current studies, the pump was independently driven by an electric motor connected to an external electrical supply. Usually the big end bearings are included in the contribution of the piston assembly.

Here, these are included in the crankshaft assembly contribution. The revised crankshaft assembly which includes 5 main bearings and 4 big end bearings account for about 1/3 of the total engine friction and it is the second largest component after the piston assembly. The valve train and ancillaries account for the remainder, contributing 12% and 8% respectively.

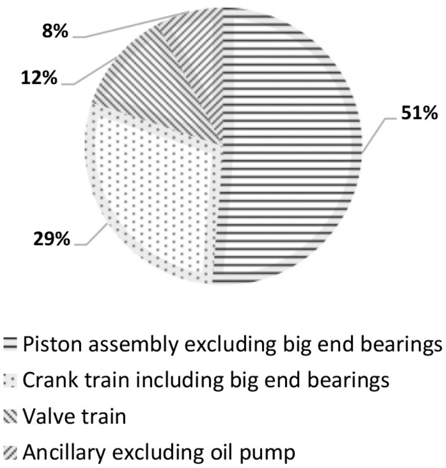

Figure 2. Distribution of engine friction contributions predicted using the model described in [1], at 1500 rpm, fully warm operating conditions.

Lubrication Circuit Modification

The standard lubricating oil system, before modifications apart from the inclusion of a flow meter, is shown in Figure 3. Pressurized oil from the pump flows through the filter-cooler assembly (FCA) and into the main gallery. There are feeds off the main gallery to the valve train and the turbocharger, and separate feeds to each main bearing and cross drillings through the crankshaft webs feeding oil to the big end bearing journals. The piston cooling jets are also fed from the main gallery. The oil pump is driven off the nose of the crankshaft.

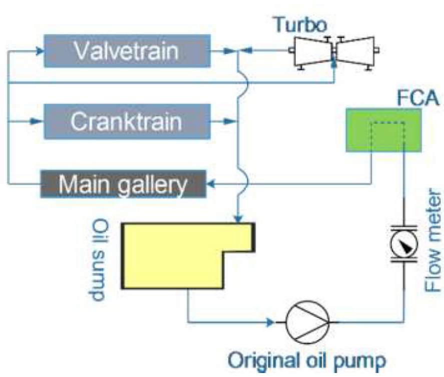

Figure 3. Original lubrication line of the engine.

The modified system is shown in Figure 5. The position and drive arrangement of the standard oil pump made the change to an electric drive difficult. The pump was replaced with a sealed unit and located off the engine where it could be driven by a mains powered electric motor. To achieve oil flow distribution control, an 'intermediate plate' was sandwiched in between the engine block and FCA (see Figure 4). Pressurised oil from the external oil pump is fed to the FCA via drillings in the plate. An external oil pipe runs parallel to engine block then directs the oil out of FCA to valve train, crank train, and turbocharger. Oil flow to crank train is controlled using servo-actuated ball valve which is located downstream of the junction where oil is directed to valve train and turbocharger. Two pressure transducers have been fitted: One is to measure the oil pressures at the 'intermediate plate'; the other one is to measure the oil pressure of the crank train which is located after the servo controlled ball valve.

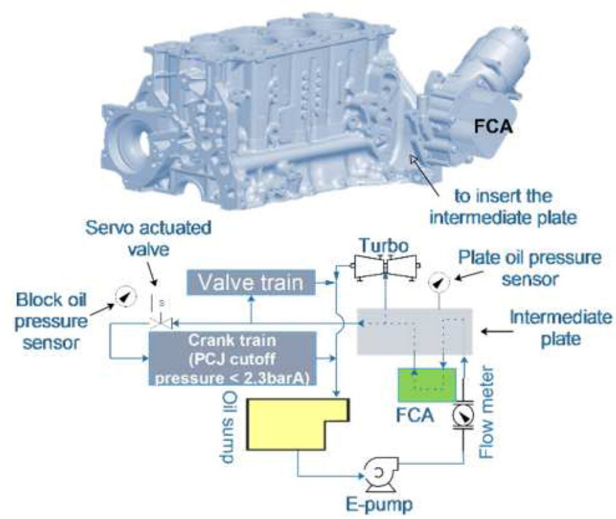

Figure 4. Modified lubrication system of the engine.

Plate oil pressure and crank train oil pressure have been used as feedback signals to feed two PI controllers (see Figure 5 and Figure 6). The controllers were tuned to achieve satisfactory control of both pressures.

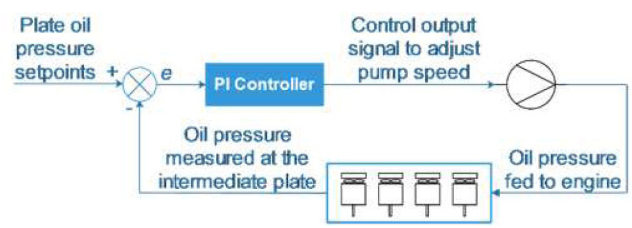

Figure 5. Closed loop control of plate oil pressure.

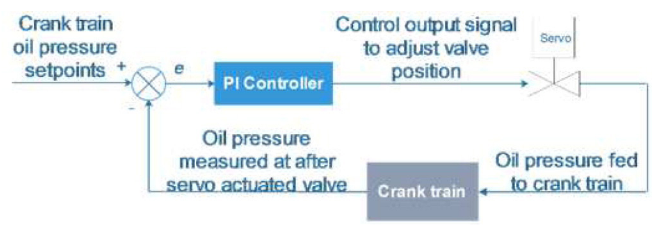

Figure 6. Closed loop control of crank train oil pressure.

The control of oil pressures is illustrated in Figure 7 for a warm-up test in which the controllers were required to maintain target pressures by adjusting the oil pump speed and ball valve position. The plate oil pressure was set to 3.5barA, and the crank train oil pressure was reduced from an initial value of 2.5barA to a target pressure of 1.5barA. The standard deviations of the plate oil pressure and crank train oil pressure during warm-up tests were 0.01barA (0.2%) and 0.032barA (2.1%) respectively.

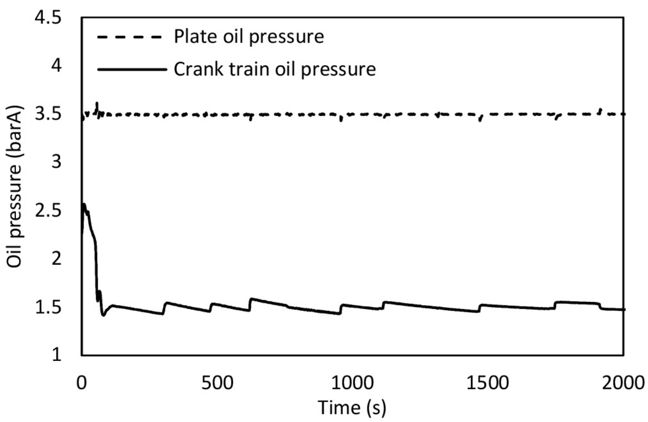

Figure 7. Plate and crank train oil pressure variations during engine warm-up.

During fully warm tests, the plate oil pressure was set at 3.3barA while crank train oil pressure follows a step change between 1.5barA and 3barA. When reducing oil pressure, the overshoot was minimized to avoid the unplanned risk of damage at the cost of a small increase in response time (see Figure 8 and Figure 9).

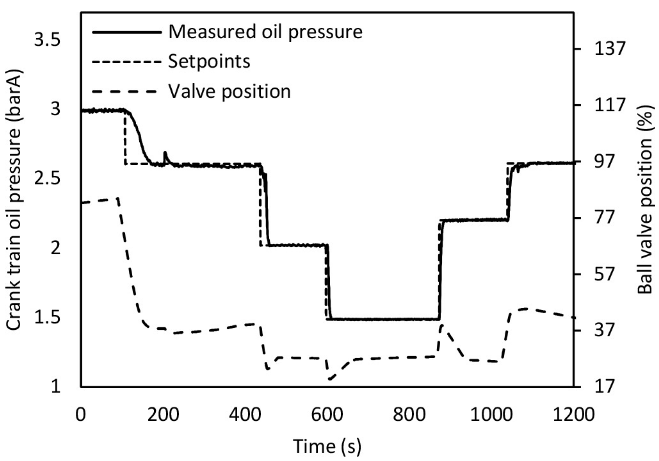

Figure 8. Crank train oil pressure step change between 1.5barA and 3barA.

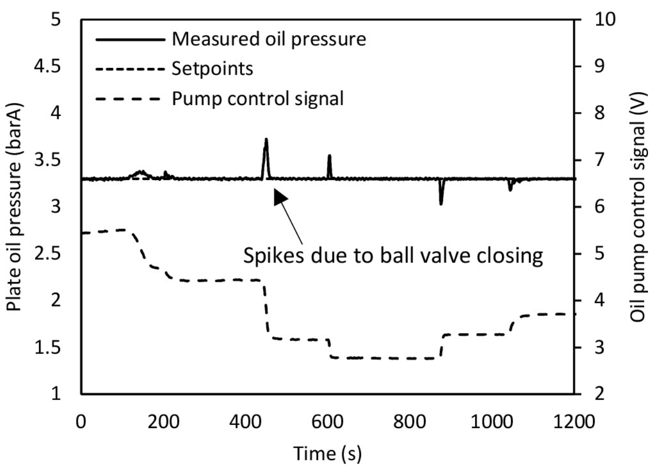

Figure 9. Plate oil pressure has been fixed at 3.3barA while crank train oil pressure follows a step change between 3barA and 1.5barA.

Experimental Results

Warm-up and fully warm tests were carried out on the engine test bed described to quantify the reduction in engine friction resulting from the reduction of oil pressure in the feed to the crank-train. The change in friction loss was determined from differences between net indicated work and brake work output. In all tests, the net indicated work was maintained constant and the changes in friction losses and ancillary work recorded through the change in brake load. If FMEP is defined as the friction work per cycle per unit displaced volume, and AMEP is the corresponding ancillary work, then Equations 1 to 3 follow [24]:

$$FMEP^{*} = FMEP + AMEP = IMEPn - BMEP$$

Equation 1

$$IMEPn = \frac{\oint_0^{720} pdV}{n * V_s}$$

Equation 2

where the integral is evaluated over a 720°CA period, and

$$BMEP = \frac{2\pi * n_R * T_b}{n * V_s}$$

Equation 3

Here we have distinguished between rubbing friction mean effective pressure, FMEP, and FMEP* which includes AMEP and is determined from the difference IMEPn - BMEP, the ancillary work term does not include work done by the oil pump. In the modified oil system, the oil pump is driven externally and does not contribute to the ancillary work. The remaining ancillary work due to the coolant pump, fuel injection pump and unloaded alternator is independent of changes in oil feed pressure. It follows that a change in the value of FMEP* when oil feed pressure is reduced is attributable just to changes in FMEP.

The engine oil used throughout the test programme was Havoline Energy 5W-30, and to pre-condition the oil the engine was run at medium speed and load for roughly 10 hours before conducting any tests. The engine had been run in before this. During the tests, engine speed was governed by a DC dynamometer and engine torque was adjusted by adjusting fuel demand using pedal control unit.

Warm-Up Tests

Two warm-up tests were carried out at 1100rpm and an IMEPn of 1.6bar at each of three crank train oil pressures: 3barA, 2barA and 1.5barA; the plate oil pressures were fixed at 3.5barA. The engine oil temperature was around 18°C when the engine was started, increasing to 80°C when engine is fully warm (at low load running conditions, oil temperature settles at 80°C once the coolant thermostat valve open on this engine). A comparison of the engine FMEP* variation during warm-up at these three crank train oil pressures is shown in Figure 10. During the warm-ups, dynamic viscosity of the oil reduced from 85mPa.s to about 11mPa.s, and FMEP* more than halved in value reflecting the dependence of rubbing friction on oil viscosity [13]. It can be seen that the FMEP* results clearly separate into 3 pairs with 3barA oil pressure FMEP* at the top, 1.5barA FMEP* at the bottom. When oil temperature is 20°C, FMEP* reduction rates are 18.6% and 10.7% after oil pressure was reduced to1.5barA from 3barA and 2barA respectively. When oil temperature is 80°C, the FMEP* reduction produced by a reduction in feed pressure is slightly lower than that when oil temperature is 20°C (see Table 2).

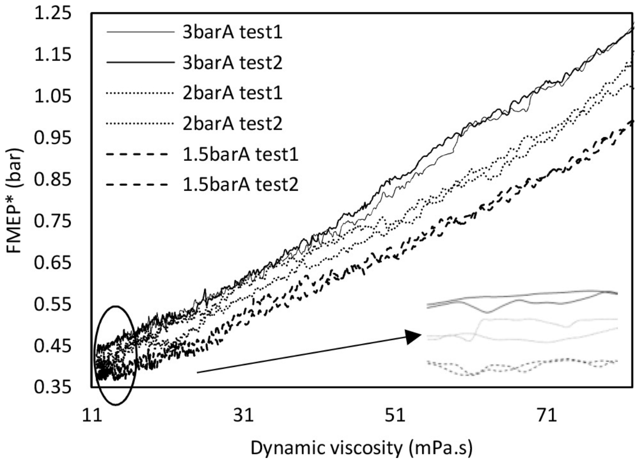

Figure 10. FMEP comparison during engine warm-up, N=1100rpm, IMEPn=1.6bar.

Table 2. FMEP reduction rate when oil is cold and hot during warm-up tests.

Crank train oil pressure change range	FMEP reduction rate when oil is 20°C (%)	FMEP reduction rate when oil is 80°C (%)
3 to 1.5barA	18.6	17.0
2 to 1.5barA	10.7	10.0

Fully Warm Tests

Tests under fully warm engine conditions were carried out over a range of loads and engine speeds. The oil film temperature was constant at around 85-90°C, so the effects of oil viscosity variation during a test was negligible. For each condition the feed pressure to the valve train and turbocharger was maintained constant by adjusting the pump delivery and the feed pressure to the crankshaft main and big end bearings was varied between 3barA and 1.5barA in step changes. This range covers the pressure of ~2.3 barA below which the piston cooling jets (PCJs) close. These jets were fed from the same oil gallery as the bearings. Potentially the effect the PCJs closing is a drop in piston temperature, which can change by tens of degrees Celsius [25], and a change in the lubrication of the piston-liner interface. Because this could produce a change in friction which might confound the measurement of friction changes due to reduced flow to the main and big-end journal bearings, the pressure at which the PCJs opened and closed marked an important point in the sweep of gallery pressure. This pressure was determined by monitoring the response of oil flow into the gallery as the feed pressure was changed. Because the PCJs take a high proportion of the total flow, switching between jets on and jets off produces a marked change in the sensitivity of flow rate to feed pressure. The opening and closing pressure of the jets is dictated by the operation of a simple spring loaded check valve within each jet.

Changing the feed pressure between 1.4barA to 3barA in increments of 0.2bar, and then reduced back to 1.4barA in the same manner produced the changes in pump flow rate required to maintain the target pressure which are plotted in Figure 11. When the feed pressure is increasing, the jets started to open at about 2.3barA and were fully open at about 2.6barA. When the feed pressure was reducing, the jets started to close at ~2.6barA, and were fully closed at about 2.3barA.

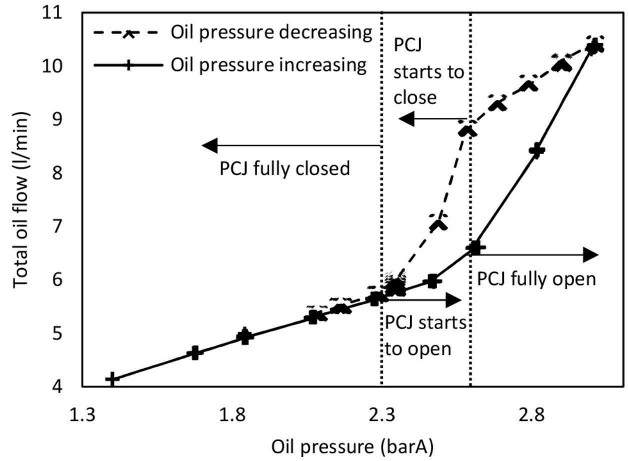

Figure 11. Influence of piston cooling jets (PCJs) opening on total oil flow.

Fully warm tests were carried out to investigate responses of engine FMEP* to changes in feed pressure to the crankshaft and PCJs. The engine speeds covered were between 1000rpm and 2000rpm, IMEPn between 1.5bar and 3.5bar. The pump delivery pressure was maintained constant at 3.3barA for all the tests. The feed pressure to the crankshaft/PCJs was increased from 1.5barA to 2.1barA, 2.5barA and 3.0barA in turn and then reduced through the same steps to 1.5barA. The pressure was settled at each pressure point for roughly 200 seconds to allow bearing oil film temperatures to adjust to new stable values. Each set of tests was repeated 3 times in order to calculate the 95% confidence interval (CI) standard errors for each test conditions (see Table 3 to Table 5 in appendix).

The test results presented in Figure 12 to Figure 14 show the average engine FMEP* values for each test condition plotted over the raw data points. The repeatability of the tests was excellent. The general trends show FMEP* increasing with engine speed and load, consistent with rubbing friction dominating the sum of FMEP and AMEP. Importantly, for any given engine load and speed, FMEP* consistently decreases as the pressure of the oil feed is reduced. The rate of decrease is almost constant and the small change between 2 and 2.5barA where the PCJs shut indicates these have not had a strong effect on FMEP. Nevertheless, in quantifying the changes in FMEP produced by reducing the oil feed pressure to 1.5barA, the benefit has been calculated relative to 2.5barA, when the piston cooling jets are just closed, and 3.0barA corresponding to standard operating pressure and when the PCJs are open.

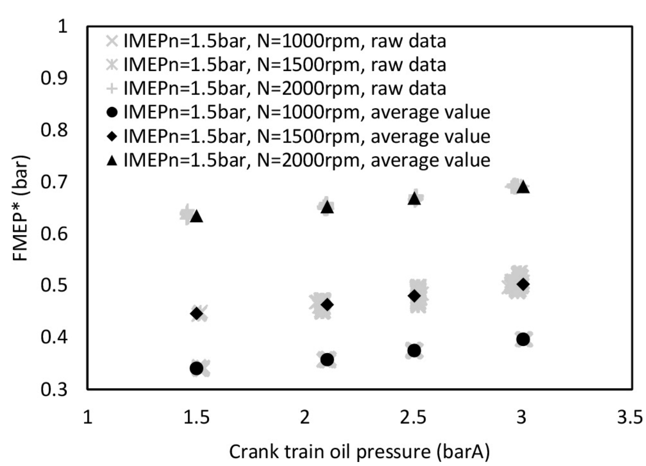

Figure 12. Fully warm tests results, IMEPn = 1.5bar, oil temperature around 90°C.

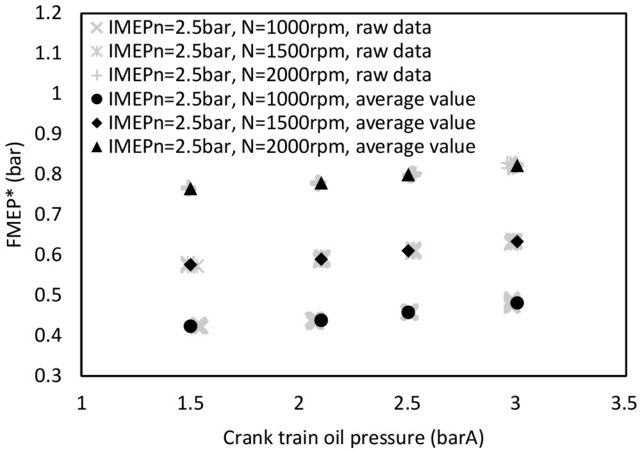

Figure 13. Fully warm tests results, IMEPn = 2.5bar, oil temperature around 90°C.

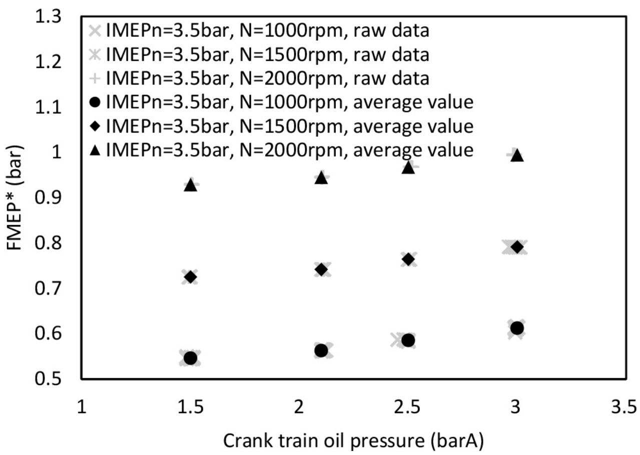

Figure 14. Fully warm tests results, IMEPn = 3.5bar, oil temperature around 90°C.

FMEP Reduction Map

Contours of the percentage reduction in FMEP produced by reducing feed pressure from 3 to 1.5barA, 2.5 to 1.5barA and 2.1 to 1.5barA are given in Figure 15 to Figure 17 respectively. Piecewise cubic spline surfaces were computed using the fully warm FMEP data. The maps cover engine speed range between 1000 and 2000rpm, and IMEPn range between 1.5 and 3.5bar.

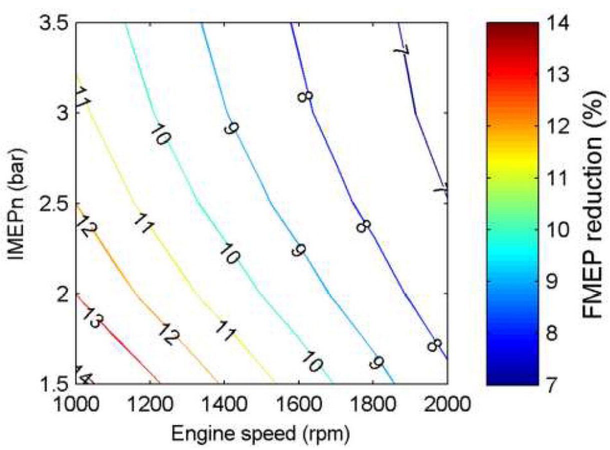

Figure 15. Engine FMEP reduction map, block oil pressure reduced from 3 to 1.5barA.

The FMEP reductions are highest at low engine speed and low load / IMEPn. This is consistent with the observations described in [26]. The highest value of 14% reduction in FMEP was achieved at 1000rpm and 1.5bar IMEPn for an oil feed pressure reduction from 3barA to 1.5barA. For the same oil pressure change range, a 9% FMEP reduction was measured at 1500rpm and 1.5bar IMEPn. This represents a reduction of 1/3 of the 28% FMEP contribution of the main bearings and big end bearings to the total engine FMEP, as shown in Figure 2.

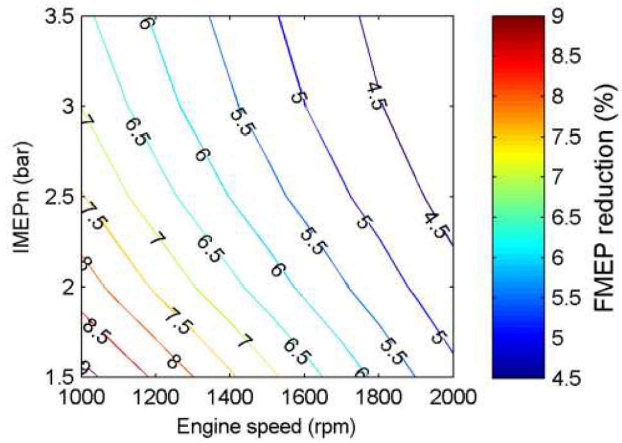

Figure 16. Engine FMEP reduction map, block oil pressure reduced from 2.5 to 1.5barA.

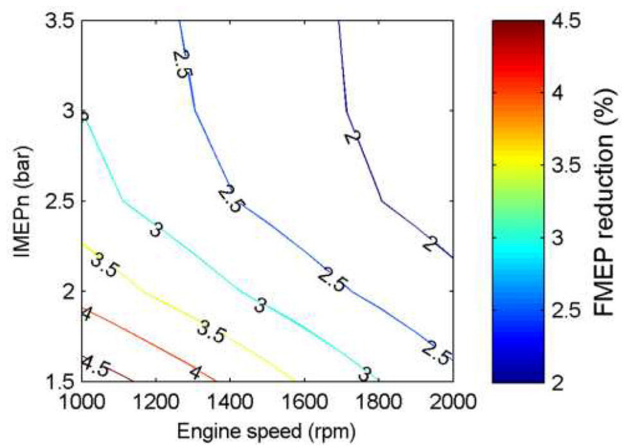

Figure 17. Engine FMEP reduction map, block oil pressure reduced from 2.1 to 1.5barA.

Discussion

Fuel Consumption Reduction over NEDC

The FMEP reduction maps developed above have been applied to estimate the fuel consumption reduction which would be achieved over the New European Drive Cycle (NEDC) cycle for a fully-warm test engine in a Ford Focus. The engine speed and BMEP profile for this combination of engine and vehicle is shown in Figure 18. The FMEP reduction maps cover most of the engine speeds in the cycle, but only BMEP values up to ~4bar. This does not cover the full range of loads imposed during the cycle, particularly during the more highly loaded Extra Urban Drive Cycle (EUDC) part when BMEP approaches 15bar for short periods. Here, noting that the FMEP reduction decreases with increasing load, the reduction was taken to be zero for BMEP values outside the mapped region. This will produce an underestimate of the fuel consumption saving.

In Figure 15 to Figure 17, the reduction in friction has been mapped as a function of IMEPn whilst the load variation during the NEDC is defined as a BMEP variation. The IMEPn corresponding to a BMEP was found from Equation 1, using the value of FMEP* determined experimentally at the relevant baseline value of oil feed pressure (either 3.0bar, 2.5bar or 2.1bar). The reductions in FMEP are relative to the FMEP values at these baseline conditions.

Figure 19 shows the transient FMEP reduction over NEDC cycle calculated using the fully warm FMEP reduction maps against BMEP. A maximum of 14%, 9%, and 5% FMEP reduction has been achieved when crank train oil pressure is reduced to 1.5barA from 3barA, 2.5barA, and 2.1barA respectively.

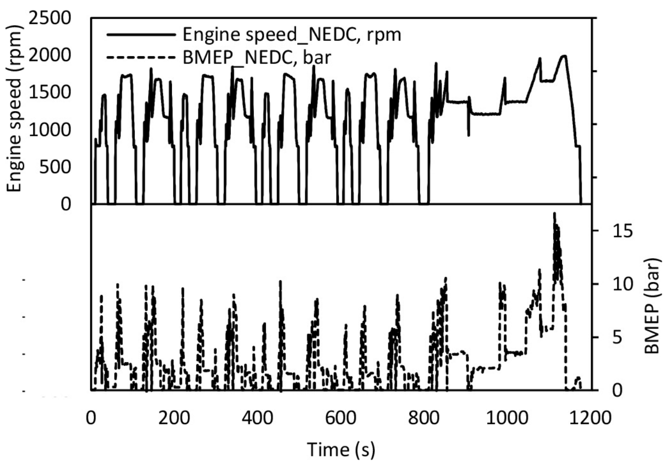

Figure 18. Engine speed and load for the test engine in a B segment car driven through the NEDC.

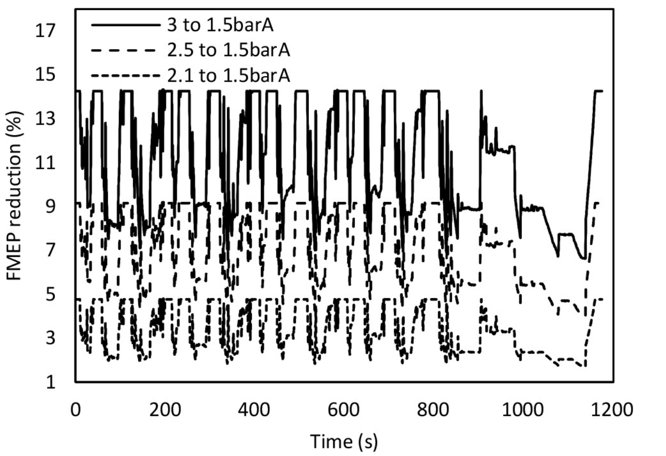

Figure 19. Transient FMEP reduction over NEDC cycle, accuracy depends on the accuracy of the fully warm FMEP reduction map.

The frictional work reductions for the Urban Drive Cycle (UDC), EUDC and the whole NEDC calculated by using Equation 4 are shown in Figure 20. When the crank train oil pressure is reduced from 3barA to 1.5barA, the reduction in frictional work over the UDC is 10.38%. This is higher than the 10% reduction for the whole NEDC cycle because the UDC has a higher proportion of low speed and low load operating points where the frictional work reduction is higher. Conversely the reduction in frictional work over the EUDC is lower at 8.9%. If a lower feed pressure is used as the baseline and the reduction in oil pressure is from 2.5barA to 1.5barA, or 2.1barA to 1.5barA, the frictional work reductions are smaller as also shown in Figure 20.

$$Frictional\ work\ reduction(\%) = 100 \frac{\int_0^{\tau} FMEP_{reduction} * n * V_s * \frac{N(rpm)}{60 \times n_R} dt}{\int_0^{\tau} FMEP^*{}_{baseline} * n * V_s * \frac{N(rpm)}{60 \times n_R} dt}$$

Equation 4

In Equation 4, τ is the duration of the cycle. For the UDC, EUDC and complete NEDC this is 780s, 400s and 1180s, respectively. Because the predictions are based on a fully warm start condition, the % reduction over the UDC is the same for each of the 4 repeated elementary cycles which make this up.

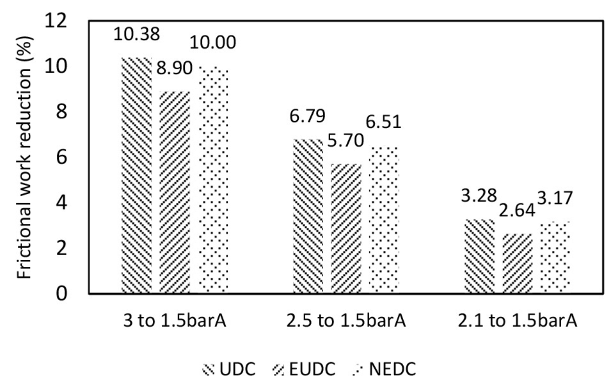

Figure 20. Frictional work reduction prediction NEDC cycle using FMEP reduction rate map over, accuracy depends on the accuracy of the fully warm FMEP reduction map.

The corresponding predictions of reductions in fuel consumption over the UDC, EUDC and NEDC are given in Figure 21. The predictions were made using Equation 5 and Equation 6 and assuming the gross indicated specific fuel consumption was constant. When crank train oil pressure is reduced from 3barA to 1.5barA, fuel consumption reduction over the UDC is 1.64% which is higher than the 1% reduction for the EUDC cycle, and 1.41% for the NEDC. When the crank train oil pressure is reduced from 2.5barA to 1.5barA, the predicted fuel consumption reductions for the UDC, EUDC, and NEDC are 1.04%, 0.62%, and 0.89% respectively. When the crank train oil pressure is reduced from 2.1barA to 1.5barA, the corresponding reductions in fuel consumption are 0.48%, 0.28%, and 0.42% respectively.

$$Fuel\ consumption\ reduction(\%) = \frac{Frictional\ work\ reduction}{Gross\ indicated\ work} = 100 \frac{\int_0^{\tau} FMEP_{reduction} * n * V_s * \frac{N(rpm)}{60 \times n_R} dt}{\int_0^{\tau} IMEP_g * n * V_s * \frac{N(rpm)}{60 \times n_R} dt}$$

Equation 5

$$IMEP_g = BMEP + PMEP + FMEP^*$$

Equation 6

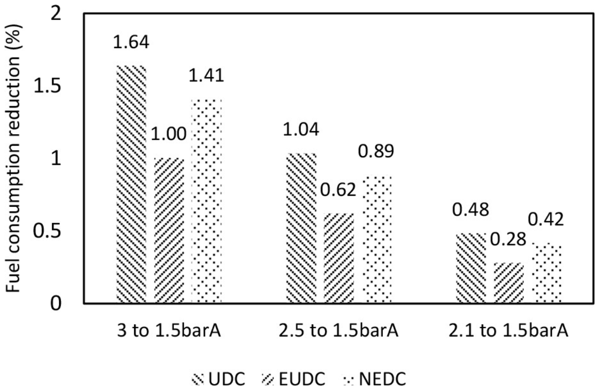

Figure 21. Fuel consumption reduction prediction over NEDC cycle calculated based on FMEP reduction map, accuracy depends on the fully warm FMEP reduction map.

Reductions in Oil Pump Work

In the set-up used in this study, the oil pump was located off the engine and driven by a mains powered electric motor. Any change in oil pump work had no effect on the AMEP of the engine or inferred changes in rubbing friction. Nevertheless, it is instructive to consider how the pump work is influenced as when the oil pump is mounted on the engine and driven off the crankshaft, this makes a large contribution to ancillary load on the engine.

The pump flow delivery was varied to maintain a target delivery pressure of 3.3barA for the feed to the valve train and the turbocharger but because all the pump flow was delivered at this pressure, only changes in flow rate result in changes in the hydraulic power required:

$$\dot{W}_h = \Delta p \times \dot{V}$$

Equation 7

Only downstream of the flow split between the feed to the valve train and turbocharger and the feed to the crankshaft and PCJs was the feed pressure for the latter adjusted. Although not changing the delivery pressure of the pump, this affected the total flow delivery. The total flow was reduced by the reduction of flow to the main and big end bearings; if the pressure change caused the PCJs to close, this would also reduce the total flow. Referring to Figure 11, when the feed pressure to the crankshaft and PCJs was 3bar, the total pump delivery was ~10.5 l/min. When the feed pressure was reduced to 1.5bar, the total flow was reduced to ~4.2 l/min, representing a 60% reduction in hydraulic work input at the pump. Of this, the closure of the PCJs accounted for around half of the flow reduction; the reduction in pump hydraulic work associated with reduced flow to the crankshaft main and big end bearings is around 30%.

The dual-pressure feature of the oil system in the form used here does not reduce pump work. The total flow from the oil pump is delivered at the same pressure and the pressure reduction for the low pressure feed takes place downstream. A reduction in work might be achieved using, for example, a pump providing dual delivery streams. The reduction in pump work associated with just the reduction in flow delivery might equally be achieved with a variable capacity pump. In this case, decoupling the pump drive from engine speed would not be necessary.

Safe Limits on Feed Pressure

The setting of the lower level on oil feed pressure for the crankshaft and big end bearings was based on limited experimental results from related work covering a range of conditions including a failure case. The lower limit on pressure and upper limit on engine load and speed were set by the boundaries of this limited experience. In the current work, no indications of stress were detected in bearing film temperatures. The engine has not been stripped to examine the bearings. The aim of the current work was to assess the potential of operating with low oil pressure under controlled conditions; in future work, clearly it would be important to establish the safe limits of operation in greater detail, and to define criteria for detecting stress before damage occurs.

Conclusion

The oil flow control system described regulates the feed pressure of the oil flow to the crankshaft main bearings, the big end bearings and the piston cooling jets at a target value while maintaining a higher constant feed pressure of the oil flow to the valve train and turbocharger.

Reducing the oil feed pressure to the five crankshaft main bearings and the four big end bearings from 3 barA to 1.5 barA whilst maintaining the feed to the valve train and turbocharger at 3barA reduced engine FMEP by up to 14% at light loads, with diminishing benefit as load was raised. The closure of the piston cooling jets at pressures below 2.3barA did not affect this FMEP reduction.

Reducing oil feed pressure from 2.1 barA to 1.5 barA with the piston cooling jets closed throughout produced reductions in engine FMEP up to 4.5%. The benefit was solely due to a reduction of bearing friction losses.

A reduction in fuel used over the NEDC of up to 1.64% is predicted for the reduced friction in the main and big end bearings.

References

1. Shayler, P., Leong, D., and Murphy, M., "Contributions to Engine Friction During Cold, Low Speed Running and the Dependence on Oil Viscosity," SAE Technical Paper 2005-01-1654, 2005, doi:10.4271/2005-01-1654.
2. Priest M. and Taylor C., "Automobile engine tribology - approaching the surface," *Wear*, pp. 193-203, 2000.
3. Taylor C., "Automobile engine tribology-design considerations for efficiency and durability," *Wear*, vol. 221, pp. 1-8, 1998.
4. Shayler P. J., Leong D., Pegg I. and Murphy M., "Investigations of Piston Ring Pack and Skirt Contributions to Motored Engine Friction," *SAE Paper* 2008-01-1046, *SAE World Congress*, Detroit, 2008; Also SAE Int. J. Engines, vol. 1, no. 1, pp. 794-803, 2008.
5. Richardson D. E., "Review of Power Cylinder Friction for Diesel Engines," *Journal of Engineering for Gas Turbines and Power*, vol. 122, pp. 506-519, 2000.
6. Shayler P., Allen A., Leong D. and Pegg I. e. a., "Characterising Lubricating Oil Viscosity to Describe Effects on Engine Friction," *JSAE Paper No* 20077202/ *SAE Paper No* 2007-01-1984, JSAE Int. Conf. Fuels and Lubricants, Kyoto, July 2007.

7. Zammit J.-P., Shayler P. J. and Pegg I., "Thermal coupling and energy flows between coolant, engine structure and lubricating oil during engine warm up," *IMechE/SAE Proceedings of VTMS 10, paper C1305-053*, Gaydon, 2011.
8. Higashitani M., Yamada K., Yoshijima K. and Kobayashi H., "Dual-Chamber Oil Pan for Improved Engine Warm-up Performance," *MTZ*, Vol 73, pp54-56, 2012.
9. Law T., Shayler P. J. and Pegg I., "Investigations of Sump Design to Improve the Thermal Management of Oil Temperature During Engine Warm-up," *IMechE Paper 640/044/2007,Proc. SAE/IMechE Int. Conf VTMS8*, Nottingham, May 2007.
10. Zammit, J., Shayler, P., Gardiner, R., and Pegg, I., "Investigating the Potential to Reduce Crankshaft Main Bearing Friction During Engine Warm-up by Raising Oil Feed Temperature," *SAE Int. J. Engines* 5(3):1312-1319, 2012, doi:10.4271/2012-01-1216.
11. Bent E., Shayler P. J., Rocca A. L. and Rouaud C., "The effectiveness of stop-start and thermal management measures to improve fuel economy," Proc. of IMechE Int Conf C1365, VTMS 11, C1365, paper C027, Coventry, 2013.
12. Gardiner R., Zhao C., Addison J. and Shayler P. J., "Investigations of Thermal State Changes during the Warm up of a Spark Ignition Engine," Proc. of IMechE Int Conf C1365, VTMS 11, paper C028, Coventry, 2013.
13. Shayler P., Baylis W. and Murphy M., "Main Bearing Friction and Thermal Interaction During the Early Seconds of Cold Engine Operation," *Journal of Engineering for Gas Turbines and Power*, pp. 197-204, 2005.
14. chisaki, J., Yoshijima, K., Kikuchi, T., Morinaka, S. et al., "Development of a New 2.0-Liter Fuel-Efficient Diesel Engine," SAE Technical Paper 2013-01-0310, 2013, doi:10.4271/2013-01-0310.
15. He M., Allaire P., Barrett L. and Nicholas J., "Thermohydrodynamic modeling of leading-edge groove bearings under starvation condition," *Tribology transactions*, vol. 48, no. 3, pp. 362-369, 2005.
16. Edney S., Heitland G. B. and Decamillo S., "Testing, Analysis, and CFD Modeling of a Profiled Leading Edge Groove Tilting Pad Journal Bearing," in International Gas Turbine & Aeroengine Congress & Exhibition, Stockholm, Sweden, 1998.
17. Heshmat H. and Pinkus O., "Performance of starved journal bearings with oil ring lubrication," in ASME/ASLE joint lubrication conference, San Diego, Calif., 1984.
18. Heshmat H., "The mechanism of cavitation in hydrodyanamic lubrication," *Tribology transcactions*, vol. 34, no. 2, pp. 177-186, 1991.
19. Tanaka M., "Jouranl bearing performance under starved lubrication," *Tribology international*, vol. 33, pp. 259-264, 2000.
20. Neukirchner H., Kramer M. and Ohnesorge T., "The controlled vane-type oil pump for oil supply on demand for passenger car engines.," in *SAE Technical paper* 2002-01-1319, 2002.
21. Ribeiro E., Melo W. and Filho A., "Application of electric oil pumps on automotive systems," in *SAE Technical Paper* 2005-01-4086, 2005.
22. Brace C., Hawley J., Cox A., Pegg I. and Stark R., "The effect of variable flow oil pumps on vehicle fuel economy," in *Vehicle fuel economy*, Government accountablility office, 2008, pp. 219-226.
23. Malvasi A., Squarcini R., Armenio G. and Brommel A., "Design Process of an Electric Powered Oil Pump," *Autotechreview*, Vol 3 Issue 3, pp36-39.
24. HeywoodJ. B., Internal Combustion Engine Fundamentals, McGraw-Hill, 1988.
25. Luff D., Law T., Shayler P. and Pegg I., "The Effect of Piston Cooling Jets on Diesel Engine Piston Temperatures, Emissions and Fuel Consumption," *SAE International Journal of Engines -V121-3*, vol. 5, no. 3, pp. 1300-1311, 2012.
26. Vijayaraghavan D., "Effect of lubricant supply starnavtion on the thermohydrodynamic performance of a jouranl bearing," *Tribology transactions*, vol. 39, no. 3, pp. 645-653, 1996.

Contact Information

Professor Paul Shayler
Department of Mechanical, Materials and Manufacturing Engineering
University of Nottingham, Nottingham NG7 2RD, UK
Paul.Shayler@nottingham.ac.uk

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Definitions/Abbreviations

BarA - Absolute pressure in bar
BarG - Gauge pressure in bar
BMEP - Brake mean effective pressure
CI - Confidence interval
EUDC - Extra urban drive cycle (Part 2 of the NEDC)
FCA - Oil filter cooler assembly
FMEP - Friction mean effective pressure
IMEPg - Gross indicated mean effective pressure
IMEPn - Net indicated mean effective pressure
NEDC - New European drive cycle
n - Number of cylinders
n_R - Number of revolutions per cycle
p - Cylinder pressure
PCJ - Piston cooling jet
PMEP - Pumping mean effective pressure
T_b - Engine brake torque
UDC - Urban drive cycle (Part 1 of the NEDC)
V - Volume
$\dot{V}$ - Volume flow rate of oil
Vs - Cylinder swept volume
$\dot{W}_h$ - Oil pump hydraulic power
τ - Length of drive cycle
Δp - Oil pressure change across pump

APPENDIX

Table 3. 95% CI Standard error analysis of fully warm test results, IMEPn=1.5bar.

Test condition		Mean FMEP (bar)	Standard error (3 repeat tests) 95% CI	
Engine speed (rpm)	Oil pressure (barA)		(+/- bar)	(+/- %)
1000	3	0.397	0.005	1.16
	2.5	0.375	0.004	1.17
	2.1	0.357	0.004	1.19
	1.5	0.340	0.004	1.22
1500	3	0.503	0.004	0.83
	2.5	0.481	0.004	0.88
	2.1	0.463	0.004	0.94
	1.5	0.446	0.005	1.02
2000	3	0.692	0.005	0.66
	2.5	0.669	0.004	0.65
	2.1	0.652	0.004	0.65
	1.5	0.635	0.004	0.66

Table 4. 95% CI Standard error analysis of fully warm test results, IMEPn=2.5bar.

Test condition		Mean FMEP (bar)	Standard error (3 repeat tests) 95% CI	
Engine speed (rpm)	Oil pressure (barA)		(+/- bar)	(+/- %)
1000	3	0.481	0.004	0.87
	2.5	0.458	0.004	0.92
	2.1	0.437	0.004	0.99
	1.5	0.423	0.005	1.07
1500	3	0.633	0.005	0.72
	2.5	0.610	0.004	0.72
	2.1	0.590	0.004	0.72
	1.5	0.575	0.004	0.73
2000	3	0.823	0.004	0.51
	2.5	0.800	0.004	0.53
	2.1	0.779	0.004	0.56
	1.5	0.765	0.004	0.52

Table 5. 95% CI Standard error analysis of fully warm test results, IMEPn=3.5bar.

Test condition		Mean FMEP (bar)	Standard error (3 repeat tests) 95% CI	
Engine speed (rpm)	Oil pressure (barA)		(+/- bar)	(+/- %)
1000	3	0.612	0.005	0.75
	2.5	0.585	0.004	0.75
	2.1	0.563	0.003	0.50
	1.5	0.547	0.002	0.45
1500	3	0.791	0.002	0.31
	2.5	0.764	0.002	0.30
	2.1	0.742	0.002	0.31
	1.5	0.725	0.002	0.26
2000	3	0.995	0.002	0.25
	2.5	0.968	0.003	0.29
	2.1	0.946	0.004	0.46
	1.5	0.929	0.005	0.49

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